Notice No.4

Rules and Regulations for the Classification of Naval Ships, January 2021

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Please note that corrigenda amends to paragraphs, Tables and Figures are not shown in their entirety.

Issue date: June 2021

Amendments to	Effective date	IACS/IMO implementation (if applicable)
Volume 1, Part 1, Chapter 2, Section 3	Corrigenda	N/A
Volume 1, Part 3, Chapter 3, Section 2	Corrigendum	N/A
Volume 1, Part 3, Chapter 5, Sections 10 & 11	Corrigendum	N/A
Volume 1, Part 4, Chapter 2, Section 3	Corrigendum	N/A
Volume 1, Part 5, Chapter 3, Section 4	Corrigenda	N/A
Volume 1, Part 5, Chapter 4, Section 3	Corrigendum	N/A
Volume 1, Part 6, Chapter 2, Sections 3 & 4	Corrigenda	N/A
Volume 1, Part 6, Chapter 3, Sections 3, 14 & 15	Corrigenda	N/A
Volume 1, Part 6, Chapter 4, Section 2	Corrigenda	N/A
Volume1, Part 7, Chapter 3, Section 2	Corrigendum	N/A
Volume 2, Part 1, Chapter 3, Section 14	Corrigendum	N/A
Volume 2, Part 2, Chapter 1, Section 3	Corrigenda	N/A
Volume 2, Part 4, Chapter 2, Section 2	Corrigendum	N/A
Volume 2, Part 9, Chapter 8, Section 5	Corrigendum	N/A

Volume 1, Part 1,
Chapter 2
Classification Regulations

■ *Section 3*
Character of Classification and Class notations

3.5 Ship type notations

3.5.6 NS(SR) and NS(SSC) vessels

For vessels that are using either the **NS(SR)** or **NS(SSC)** ship type notations, the following requirements are to be complied with:

(d) Plans of systems within the scope of classification, as categorised in accordance with *Vol 2, Pt 1, Ch 1, 3.1 Categories* and not covered by the ~~Rules for Special Service Craft~~ *Rules and Regulations for the Classification of Special Service Craft* or ~~Rules for Ships~~ *Rules and Regulations for the Classification of Ships*, as appropriate, are to be submitted to LR for approval in accordance with the requirements of the respective requirements of these Rules. The following are examples of such systems:

- ii. Chilled water systems (*see* ~~*Vol 2, Pt 7, Ch 2 Ship Piping Systems*~~ *Vol 2, Pt 7, Ch 5 Ship Type Piping Systems*).
- iii. High pressure sea-water systems (*see* ~~*Vol 2, Pt 7, Ch 2 Ship Piping Systems*~~ *Vol 2, Pt 7, Ch 5 Ship Type Piping Systems*).
- iv. High and low pressure compressed air systems (*see* ~~*Vol 2, Pt 7, Ch 2 Ship Piping Systems*~~ *Vol 2, Pt 7, Ch 5 Ship Type Piping Systems*).
- v. Hydraulic power actuating systems (*see* ~~*Vol 2, Pt 7, Ch 2 Ship Piping Systems*~~ *Vol 2, Pt 7, Ch 5 Ship Type Piping Systems*).

Volume 1, Part 3,
Chapter 5
Anchoring, Mooring, Towing, Berthing, Launching, Recovery and Docking

■ *Section 10*
Anchoring equipment in deep and unsheltered water

10.3 Anchor windlass and chain stopper

Table 5.10.1 Anchoring equipment for ships in unsheltered water with depth up to 120 m

Equipment Number EN_1		High holding power stockless bower anchors		Stud link chain cable for bower anchors		
					Min. diameter	
Equal to or greater than	Less than	Number	Mass per anchor (kg)	Length (m)	Special quality (Grade U2) (mm)	Extra special quality (Grade U3) (mm)
… 8400 …	8900	2	~~2800~~ 28000	797,5	158	127

■ *Section 11*
Launch and recovery, berthing and dry-docking arrangements

11.4 Dry-docking loads

11.4.6 The following equation may be used to calculate the dry-docking load distribution, F_{-DL}, between main transverse bulkheads acting on a keel block:

f_{bhd} = ~~2~~ 0,5, for the keel blocks located adjacent to a main transverse bulkhead

Volume 1, Part 4,
Chapter 2
Military Load Specification

■ *Section 3*
Internal blast

3.5 Quasi static pressure

3.5.3 The QSP can be determined from the following:

~~P_{qs} = 2,25 (W_e N)^{0,72} x10^3 kN/m^2~~ P_{qs} = 2,25 (W_e V/)^{0,72} x10^3 kN/m^2

Volume 1, Part 5,
Chapter 3
Local Design Loads

■ *Section 4*
Impact loads on external plating

4.2 Bottom impact pressure, *IP* bi

4.2.1 The bottom impact pressure due to slamming, IP_{bi}, is to be derived using the method given below. This method will produce impact pressures over the whole of the underwater plating region:

….

where

V_{bs} = slamming velocity, in m/s, and is given by

~~$= \sqrt{V_{th} + 2m_1 IN(N_{s1})}$ for $N_{s1} \geq 1$~~ $\sqrt{{V_{\mathrm{th}}}^2 + 2m_1 ln(N_{s1})}$ for $N_{s1} \geq 1$

= 0 for $N_{s1} < 1$

4.3 Bow flare and wave impact pressures, *IP*bf

4.3.1 This Section is applicable to:

(a) Bow flare region.

(b) Sides and undersides of sponsons.

(c) Other parts of the side shell plating close to and above the design waterline that are expected to be subjected to wave impact pressures.

The bow flare wave impact pressure, wave impact pressure on sponsons and other parts of the side shell plating above the design waterline, IP_{bf}, in kN/m^2, due to relative motion is to be taken as:

….

where

V_{bf} = wave impact velocity, in m/s, and is given by

~~$= \sqrt{{V_{thbf}}^2 + 2m_1 IN(N_{bf})}$ for $N_{bf} > 1$ for $N_{bf} \geq 1$~~ $\sqrt{{V_{\mathrm{thbf}}}^2 + 2m_1 ln(N_{\mathrm{bf}})}$ for $N_{bf} \geq 1$

= 0 for $N_{bf} < 1$

Volume 1, Part 5,
Chapter 4
Global Design Loads

■ *Section 3*
Global hull girder loads

3.3 Vertical wave bending moments

3.3.1 The minimum value of vertical wave bending moment, M_{-W} at any position along the ship may be taken as follows:
$M_W = F_f D_f M_o$ kNm

An area ratio value of 1,0 results in a sagging correction of factor -1,10.
(a) ~~FfH~~ F_{fH} is the hogging (positive) moment correction factor and is to be taken as

3.3.2 The area ratio factor, ~~R_A~~ R_A, for the combined stern and bow shape is to be derived as follows:

3.3.3 The bow flare area, ~~A_{BF}~~ A_{BF}, is illustrated in *Figure 4.3.1 Deviation of bow and stern flare areas* and may be derived as follows:
$T_{C,U}$ = ~~is~~ a waterline taken $L_{-f}/2$ m above the design draught
$T_{C,U}$ = $T + L_f/2$ m
where

b_0 = projection of ~~$T_{C,U}$~~ $T_{C,U}$ waterline outboard of the design draught waterline at the FP, in metres, *see Figure 4.3.1 Deviation of bow and stern flare areas*

b_1 = projection of ~~$T_{C,U}$~~ $T_{C,U}$ waterline outboard of the design draught waterline at $0{,}9L_R$ from the AP, in metres

b_2 = projection of ~~$T_{C,U}$~~ $T_{C,U}$ waterline outboard of the design draught waterline at $0{,}8L_R$ from the AP, in metres

3.3.4 The stern flare area, ~~A_{SF}~~ A_{SF}, is illustrated in *Figure 4.3.1 Deviation of bow and stern flare areas* and is to be derived as follows:
$T_{C,L}$ = is a waterline taken $L_f/2$ m below the design draught
$T_{C,L}$ = ~~$T - L_f$ m ^{2}~~ $T - L_f/2$ m.

3.3.8 The sagging correction factor, ~~f_{fS}~~ f_{fS}, in the vertical wave bending moment formulation in *Vol 1, Pt 5, Ch 4, 3.3 Vertical wave bending moments 3.3.1* may be derived by direct calculation methods. Appropriate direct calculation methods

3.8 Bow flare impact global loads

3.8.1 The requirements of this section are applicable to fast ships operating in the displacement mode that satisfy the following requirements:
a) speed V_{sp} > 17,5 knots

b) bow shape factor ψ > 0,15

Volume 1, Part 6,
Chapter 2
Design Tools

■ *Section 3* **Buckling**

3.3 Plate panel buckling requirements

Table 2.3.1 Plate panel buckling requirements

	Stress field	Buckling interaction formula
(c)	bi-axial compressive loads	for A_R = 1,0 for other aspect ratios, i.e. $A_R \neq 1{,}0$ when *G* is taken from *Figure 2.3.3 Interaction limiting stress curves of G for plate panels subject to bi-axial compression*~~, see Table 2.3.2(c)~~

Figure 2.3.3 Interaction limiting stress curves of G for plate panels subject to bi-axial compression~~, see Table 2.3.2(c)~~

Table 2.3.3 Buckling stress of secondary stiffeners

$\sigma_{\text{-ep}}$ = elastic ~~critical~~ buckling stress, σ_e, in N/mm^2, of the supporting plate derived from ~~*Table 2.4.1 First mode of vibration constant K i*~~ *Table 2.3.2 Buckling stress of plate panels, (a) i)*

■ *Section 4* **Vibration control**

4.4 Natural frequency of plate and stiffener combination

4.4.1 The natural frequency of a plate stiffener of constant cross-section in air is given by the following:

$$\cancel{f_{air} = \frac{k_t}{20\pi l_{bz}}\sqrt{\frac{EI}{m\left(1+\frac{\pi_2 EI}{10_4 l_{bz} GA}\right)}}\text{Hz}}$$

$$f_{\text{air}} = \frac{k_{\text{i}}}{20\pi l_{\text{b}}^2}\sqrt{\frac{EI}{m\left(1+\frac{\pi^2 EI}{10^4 l_{\text{b}}^2 \text{GA}}\right)}}\,\text{Hz}$$

Volume 1, Part 6,
Chapter 3
Scantling Determination

■ *Section 3* **NS1 scantling determination**

3.10 Shell envelope framing

Table 3.3.3 Shell envelop framing forward and aft

Symbols	
~~F_S~~	~~= fatigue factor for side longitudinals for built symmetric sections, flat bars, bulbs and T bars:~~ ~~= 1,05 at keel, 1,1 at *T*, 1,0 at 1,6*T* and above~~ ~~For angle bars:~~ ~~$= 0{,}5\left(1+\frac{1{,}1}{k_s}\right)$ at keel~~ ~~$= \frac{1{,}1}{k_s}$ at $\frac{D}{2}$~~ ~~= 1,0 at 1,6 *T* and abovebuilt asymmetric sections will be specially considered.~~ ~~Intermediate values by linear interpolation~~

***F*s**	= fatigue factor for side longitudinals For built symmetric sections, flat bars, bulbs and *T* bars: = 1,05 at keel, 1,1 at *T*, 1,0 at 1,6*T* and above For angle bars: = $0,5\left(1+\frac{1,1}{k_s}\right)$ at keel = $\frac{1,1}{k_s}$ at $\frac{D}{2}$ = 1,0 at 1,6 *T* and above Intermediate values by linear interpolation Built asymmetric sections will be specially considered.

Table 3.3.4 Shell envelope primary structure

Item and location	Modulus, in cm^3
Transverse framing system:	
~~(4)~~ (4) ≠ Side stringers in dry spaces	
≠ (5) Side stringers in deep tanks	$Z = 9{,}4K_s\, S\, h\, 4l_e^2$ or as ~~(5)~~ (4) above, whichever is the greater
(6) Web frames in dry spaces above 1,6*T* (*see* Note 2)	~~$Z = C_3\, k_s\, STH\, D$~~ $Z = C_3\, k_s\, STH\sqrt{\boldsymbol{D}}$

Table 3.3.6 Deck plating

Location	Minimum thickness, in mm, *see also* Vol 1, Pt 6, Ch 3, 2.2 Corrosion margin	
	Longitudinal framing	Transverse framing
(3) Lower decks (a) effective (continuous) (b) non effective	$t = 0{,}011 s_1\sqrt{k_s}$ ~~$t = 0{,}009 s_1\sqrt{k_s}$~~ $t = 0{,}009 s_1\sqrt{k_s}$	
(4) Strength deck (a) forward of 0,925LR And aft of 0,075LR (b) Lower decks	$t = (5{,}0 + 0{,}018 L_R)\sqrt{\frac{k_s s_1}{s_b}}$ ~~$t = 0{,}009 s_1\sqrt{k_s}$~~ $t = 0{,}009 s_1\sqrt{k_s}$	

■ *Section 14*

Strengthening for bottom slamming

14.2 Strengthening of bottom forward

Table 3.14.1 Additional strengthening of bottom forward

Item	Requirements
(2) Bottom longitudinals – other than flat bars	$\cancel{Z \geq 6{,}8 \times 10^{-6} h_s s k_s \left[(17{,}5_{ls})^2 - (0{,}01s)^2 + d_W c\left(S - \frac{s}{2000}\right)\right] cm^3}$ $Z \geq 6{,}8 \times 10^{-6} h_s s k_s \left[(17{,}5 l_s)^2 - (0{,}01 l_s)^2 + d_W c\left(S - \frac{s}{2000}\right)\right]$ cm^3

■ *Section 15*

Strengthening for wave impact loads above waterline

15.3 Strengthening against wave impact loads

Table 3.15.1 Buckling procedure for primary member web plating and web stiffener

Steps	Members	
	Primary member web plating	Primary member web stiffener
Determination of the elastic critical buckling stress, σ_e, in compression, N/mm^2 (kgf/mm^2)	$\cancel{\sigma_e = \frac{9{,}87 EI_W}{1_W A_W{}^2}}$ $\sigma_e = \frac{9{,}87 EI_W}{1_W^2 A_W}$	$\cancel{\sigma_e = \frac{9{,}87 EI_S}{1_S A_S{}^2}}$ $\sigma_e = \frac{9{,}87 EI_S}{1_S^2 A_S}$
Determination of the corrected critical buckling stress, σ_{cr}, in compression, N/mm^2 (kgf/mm^2)	$\cancel{\sigma_{cr} = \sigma_0\left(1 - \frac{\sigma_0}{\sigma_e}\right)}$ $\sigma_{cr} = \sigma_0\left(1 - \frac{\sigma_0}{4\sigma_e}\right)$	where $\sigma_e > \frac{\sigma_0}{2}$

Volume 1, Part 6,
Chapter 4
Hull Girder Strength

■ *Section 2*
Hull girder strength

2.3 Shear strength

Table 4.2.2 *k* i factors

k_i factors
…
Member 3
~~$k_3 = -0{,}01\frac{A_3}{A_4} + 0{,}25$~~
$k_3 = 0{,}01\frac{A_3}{A_4} + 0{,}25$
Member 4
~~$k_4 = 0{,}01\frac{A_3}{A_4} + 0{,}25$~~
$k_4 = -0{,}01\frac{A_3}{A_4} + 0{,}25$
…

2.5 Super structures global strength

2.5.3 The design stress due to hull girder bending, σ_{hg}, in the uppermost effective tier at side may be derived according to the following formula:

~~$\sigma_{hg} = \frac{\eta_s M_R}{1000 Z_s}$ N/mm2~~ $\sigma_{hg} = \frac{\eta_s M_R}{1000 Z_s}$ N/mm^2

where

M_R = hull girder bending moment at amidships due to sagging as determined in, ~~*Vol 1, Pt 5, Ch 4, 5 Residual strength hull girder loads*~~ *Vol 1, Pt 5, Ch 4, 3 Global hull girder loads*~~,~~ in kNm

Volume 1, Part 7,
Chapter 3
Total Load Assessment, TLA

■ *Section 2*
Structural resistance

2.2 Stresses in plating

2.2.2 The bending stress in a plate panel between stiffeners due to a uniform lateral pressure is to be calculated as follows:

~~$\sigma_b = p\left(\frac{22{,}4 s\gamma\beta}{100 t_p}\right) N/mm^2$~~

$$\sigma_b = p\left(\frac{22{,}4 s\gamma\beta}{1000 t_p}\right)^2 \text{N/mm}^2$$

Volume 2, Part 1,
Chapter 3
Requirements for Design, Construction, Installation and Sea Trials of Engineering Systems

■ *Section 14*
Thrusters

14.1 Design and construction

14.1.1 For details of design and construction requirements, *see* ~~*Vol 2, Pt 4, Ch 3 Water Jet Systems*~~ *Vol 2, Pt 4, Ch 3 Thrusters.*

Volume 2, Part 2
Chapter 1
Reciprocating Internal Combustion Engines

■ *Section 3*
Crankshaft Design

3.2 Scope

3.2.9 Further information and guidance on crankshaft design is provided in the LR~~'s~~ *Guidance Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*.

3.3 Information to be submitted

3.3.2 The following information is also required for appraisal of the crankshaft (not contained in Form 2073):

- every surface treatment affecting fillets or oil holes shall be specified so as to enable calculation according to Chapter ~~2~~ 3 of the LR *Guidance Notes for* ~~*Crankshaft SCF Calculation using Finite Element Method*~~ *the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*;
    - o this is to include ~~Crankshaft~~ crankshaft fatigue enhancement factors ~~K_1~~ K_1 and ~~K_2~~ K_2 where applicable.

3.4 Symbols

3.4.1 For the purposes of this Chapter the following symbols apply, *see also*;

D_A = the outside diameter of we or twice the minimum distance between ~~centre line~~ centreline of journals and outer contour of web, whichever is less, in mm

W_{eqw} = ~~xection~~ section modulus related to cross-section of web, in mm^3

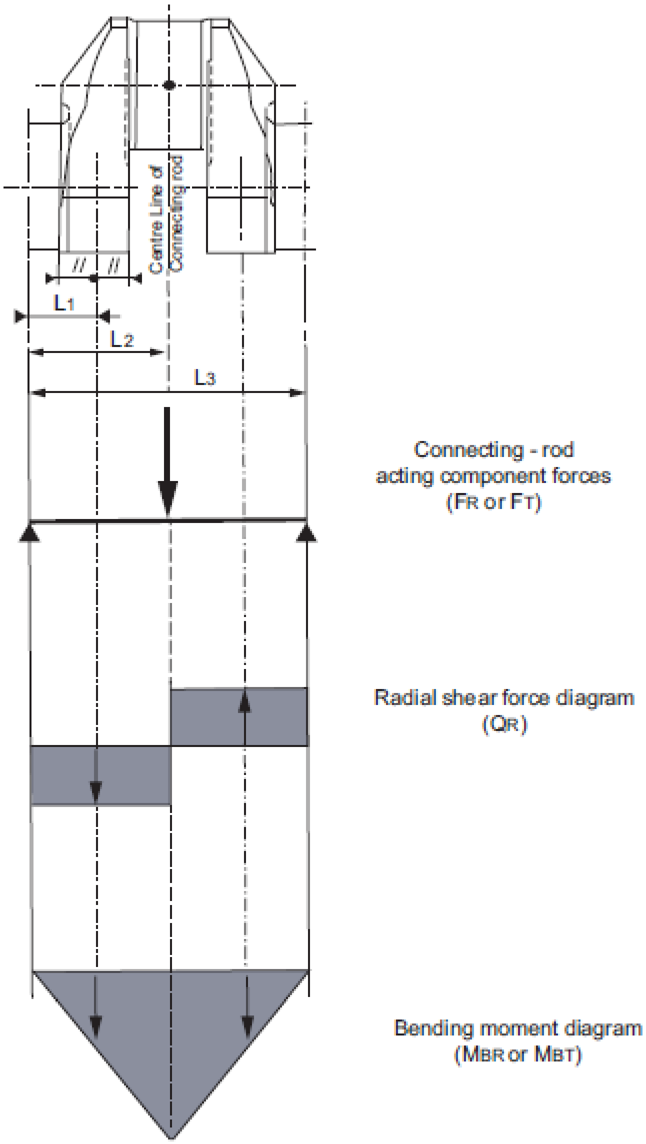
y = distance between the adjacent generating lines of journal and pin, in mm

Note for $y \geq 0,5D_s$ ~~where,w~~. Where y is less than $0,1D_s$, special consideration is to be given to the effect of the stress due to the shrink fit on the fatigue strength at the crankpin fillet.

σ_y = equivalent alternating stress for crankpin fillet, journal fillet or outlet of crankpin oil bore as applicable, in ~~N/mm^3~~ N/mm^2

3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions

Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows has been replaced with the below figure;



Crankthrow for inline engine

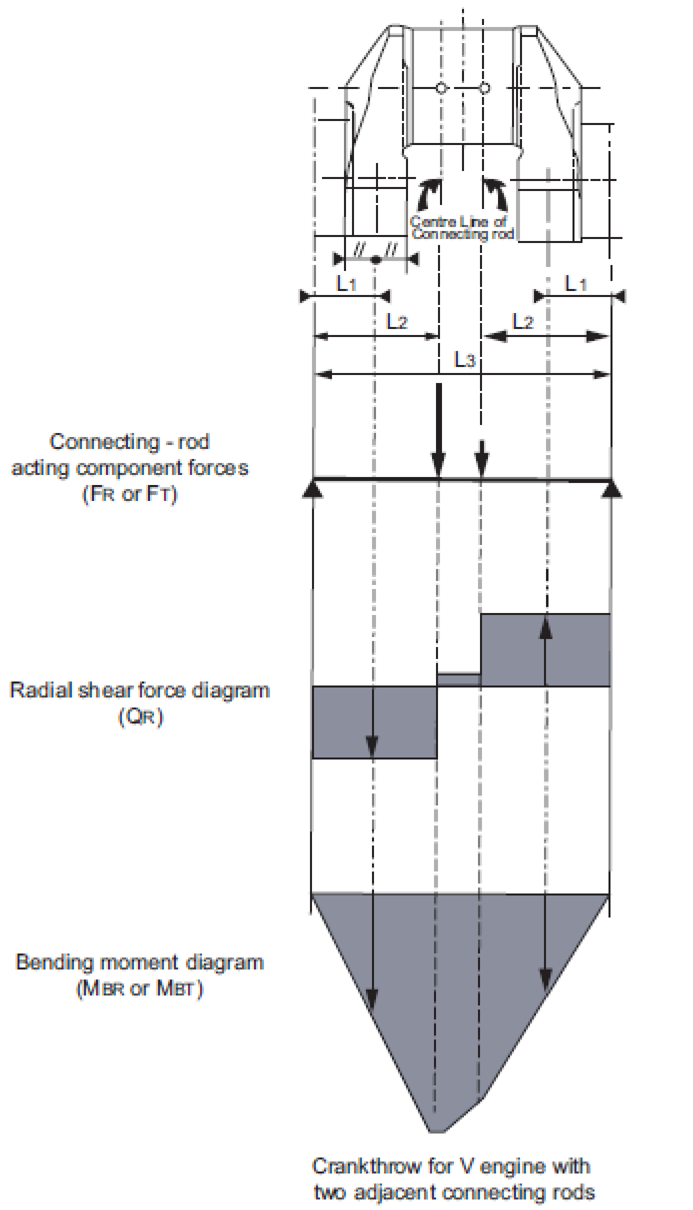
L_1 = Distance between main journal centreline and crankweb centre

L_2 = Distance between main journal centreline and connecting - rod centre

L_3 = Distance between two adjacent main journal centrelines

Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows

Figure 1.3.10 Bending moment and shear force for V engine crankthrows has been replaced with the below figure;



L_1 = Distance between main journal centreline and crankweb centre

L_2 = Distance between main journal centreline and connecting - rod centre

L_3 = Distance between two adjacent main journal centrelines

Figure 2.3.10 Bending moment and shear force for V engine crankthrows

3.5.4 ~~The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. *See Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows* and *Figure 1.3.10 Bending moment and shear force for V engine crankthrows*. M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected~~

The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. *See;*

- *Figure 1.3.5 Crankpin section through the oil bore*,
- *Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows*, and
- *Figure 1.3.10 Bending moment and shear force for V engine crankthrows*.

M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected.

3.6 Calculation of bending stresses

3.6.1 The radial and tangential forces due to gas and inertia loads acting on upon the crankpin at each connecting-rod position are to be calculated over one working cycle. Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments, M_{BRF}, M_{BRO} and M_{BTO}, and radial forces, Q_{RF}, as defined in *Vol 2, Pt 2, Ch 1 Calculation of alternating stresses due to bending moments and radial forces – assumptions ~~3.5.2~~ 3.5.3* and *Vol 2, Pt 2, Ch 1 Calculation of alternating stresses due to bending moments and radial forces – assumptions ~~3.5.3~~ 3.5.4* are then calculated.

3.6.6 Nominal alternating bending and compressive stresses in a web cross-section are calculated as follows:

$$\sigma_{BFN} = \pm\frac{M_{BRFN}}{W_{eqw}}10^3 \; K_e \text{ N/mm}^2$$

$$\sigma_{QFN} = \pm\frac{Q_{RFN}}{F} \; K_e \text{ N/mm}^2$$

where

$\cancel{M_{BRFN} = \pm\frac{1}{2}(X_{BRF\,Max} - X_{BRF\,Min})\text{Nm}}$ $M_{BRFN} = \pm\frac{1}{2}(M_{BRF\,Max} - M_{BRF\,Min})\text{Nm}$

$\cancel{Q_{RFN} = \pm\frac{1}{2}(Q_{RF\,Max} - Q_{RF\,Min})\text{Nm}}$ $Q_{RFN} = \pm\frac{1}{2}(Q_{RF\,Max} - Q_{RF\,Min})\text{N}$

3.6.7 Nominal alternating bending stress in the outlet of the crankpin oil bore is calculated as follows:

$$\sigma_{BON} = \pm\frac{M_{BON}}{W_e}10^3 \text{ N/mm}^2$$

where

M_{BON} is taken as the ½ range value $M_{BON} = \pm$ ½ ($M_{BOmax} - M_{BOMin}$) Nm

3.7 Calculation of torsional stresses

3.7.1 The nominal alternating torsional stress, $\cancel{\tau_{H}}$ τ_N, is to be taken into consideration. The value is to be derived from forced-damped vibration calculations of the complete dynamic system. Alternative methods will be given consideration. The engine designer is to advise the maximum level of alternating vibratory stress that is permitted (τ_a).

3.7.2 $\cancel{\tau a}$ τ_a or $\cancel{\tau n}$ τ_N(as applicable) is to be applied as a limiting value for the torsional vibration assessment required by *Vol 2, Pt 5, Ch 1 Torsional vibration*.

3.7.3 Nominal alternating torsional stress is calculated as follows:

$$\cancel{\tau_{H}}\ \tau_N = \frac{M_{TN}}{W_P}10^3 \text{ N/mm}^2$$

where

~~$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_G}\right)$ mm^3 for the jorunal~~

$W_p = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_p = \frac{\pi}{16}\left(\frac{D_G{}^4 - D_{BG}{}^4}{D_c}\right)$ mm^3 for the journal

3.8 Stress concentration factors

3.8.3 Where the geometry of the crankshaft is outside the boundaries (*see Table 1.3.2 Crankshaft variable boundaries for analytical SCF calculation*) of the analytical SCG the calculation method detailed in Chapter 1 and Chapter 4 of the LR *Guidance Notes for ~~Crankshaft SCG Calculation using Finite Element Method~~ Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts* may be undertaken.

3.8.5 Chapters1 and ~~3~~ 4 of the LR *Guidance Notes for ~~Crankshaft SCG Calculation using Finite Element Method~~ Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts* describe how finite element (FE) analyses can be used for the calculation of the SCF. Care needs to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

3.8.6 Crankpin SCF are calculated as follows:

a) Bending

~~$\alpha_B = 2{,}6914 f(s,w).f(w).f(b).f(r).f(d_G).f(d_H).f(\text{recess})$~~

$\alpha_B = 2{,}6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(\text{recess})$

b) Torsion

~~$\alpha_T = 0{,}8 f(r,s).f(b).f(w)$~~

$\alpha_T = 0{,}8 f \cdot (r,s) \cdot f(b) \cdot f(w)$

where
~~$f(b) = 7{,}8955 - 10{,}654b + 5{,}3482b^2 – 0{,}857b^3$~~ ~~$f(w) = w^{(-0{,}145)}$~~
$f(w) = w^{(-0{,}145)}$

3.8.7 Journal fillet SCF are calculated as follows (not applicable to semi-built) crankshafts):

a) Bending
~~$\beta_B = 2{,}7146 f_B(s,w).f_B(w).f_B(b).f_B(r).f_B(d_G).f_B(d_H).f(\text{recess})$~~
$\beta_B = 2{,}7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(\text{recess})$

b) Compression due to the radial force~~:~~
~~$\beta_Q = 3{,}0128 f_Q(s).f_Q(w).f_Q(b).f_Q(r).f_Q(d_H).f(\text{recess})$~~
$\beta_Q = 3{,}0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})$

3.9 Additional bending stress

3.9.1 In addition to the alternating bending stresses in fillets (*seeVol 2, Pt 2, Ch 1, 3.6 Calculation of bending stresses 3.6.8*) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying ~~σ_{add}~~ σ_{add} as given by *Table 1.3.3 Additional bending stresses*.

3.10 Equivalent alternating stress

3.10.4 Equivalent alternating stress, σ_V, is defined as:

(a) For the crankpin fillet:

~~$\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G^2}$ N/mm²~~ $\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H^2}$ N/mm³

(b) For the journal fillet:

~~$\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H^2}$ N/mm³~~ $\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G^2}$ N/mm²

3.11 Fatigue strength

3.11.1 The fatigue strength is to be understood as the value of equivalent alternating stress (~~Von~~ von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength can be evaluated by means of the following formulae:

~~$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right]$ N/mm²~~

$$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right] \text{ N/mm}^2$$

b) Related to the journal diameter:

~~$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D_G^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_G}}\right]$ N/mm²~~

$$\sigma_{DW} = \pm K(0{,}42\sigma_B + 39{,}3)\left[0{,}264 + 1{,}073D^{-0{,}2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_X}}\right] \text{ N/mm}^2$$

A value for K_2 will be assigned upon application by the engine designers. Full details of the process, together with the results of full scale fatigue tests will be required to be submitted for consideration. *See Chapter 2 of the* LR *Guidance ~~note - Guidance for the evaluation of Crankshaft Fatigue Tests~~ Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*.

3.11.3 Fatigue strength calculations or, alternatively, fatigue test results determined by experiment based either on full size crankthrow (or crankshaft), or on specimens taken from a full size crankthrow, may be required to demonstrate acceptability. The experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment are to be submitted for approval by LR. The procedure is to include as a minimum: method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, and confidence number. *See also* Chapter 2 of the LR *Guidance ~~for the Evaluation of Crankshaft Fatigue Tests~~ Notes for the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*. Alternatively, the following values may be taken (surface hardened zone to include fillet radii):

3.11.5 Only surface treatment processes approved by LR are permitted. Guidance for calculation of surface treated fillets and oil bore outlets is presented in Chapter ~~2~~ 3 of the LR *Guidance Notes for ~~Crankshaft SCF Calculation using Finite Element Method~~ the Calculation of Stress Concentration Factors, Fatigue Enhancement Methods and Evaluation of Fatigue Tests for Crankshafts*.

3.13 Shrink fit of semi-built crankshafts

3.13.5 The actual oversize Z of the shrink fit must be within the limits Z_{min} and Z_{max} calculated in accordance *Vol 2, Pt 2, Ch 1, 3.13 Shrink fit of semi-built crankshafts ~~3.13.7~~ 3.13.6* and *Vol 2, Pt 2, Ch 1, 3.13 Shrink fit of semi-built crankshafts ~~3.13.8~~ 3.13.7*. When *Vol 2 Pt 2, Ch 1, 3.13 Shrink fit of semi-built crankshafts 3.13.4* cannot be complied with, then the calculated values of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems. In such cases Z_{min} and Z_{max} are to be established from FEM calculations.

3.13.6 The minimum required diametral ~~Interference~~ interference is to be taken as the greater of:

$$Z_{min} \geq \frac{\sigma_{SW} D_S}{E_m} \ \text{mm}$$

and

$$Z_{min} \geq \frac{4000}{\mu\pi} \frac{S_R M_{max}}{E_m D_S L_S} \frac{1 - Q_A{}^2 Q_S{}^2}{\left(1 - Q_A{}^2\right)\left(1 - Q_S{}^2\right)} \ \text{mm}$$

where

Q_S = ~~web~~ shaft ratio, $Q_S = \frac{D_{BG}}{D_S}$

Volume 2, Part 4
Chapter 2
Water Jet Systems

■ *Section 2*
General requirements

2.1 Water jet arrangement

2.1.1 In general, for a ship to be assigned an unrestricted service notation, a minimum of two water jet systems is to be provided where these form the sole means of propulsion. For ships where a single water jet system is the sole means of propulsion or steering, a detailed engineering and safety justification is to be evaluated by LR, *see Vol 2, Pt 4, Ch 2, 2.3 Calculations and information ~~2.3.22~~ 2.3.23*. This evaluation process will include a Risk Assessment (RA) in accordance with *Vol 2, Pt 1, Ch 3, 18 Risk Assessment (RA)*, to verify that sufficient levels of redundancy and monitoring are incorporated in the water jet unit's support systems and operating equipment.

Volume 2, Part 9
Chapter 8
Programmable Electronic Systems

■ *Section 5*
Programmable electronic systems (PES)

5.1 General requirements

5.1.21 Software lifecycle activities, e.g. design, development, supply and maintenance, are to be carried out in accordance with an acceptable quality management system. Project specific software quality plans are to be submitted. These are to demonstrate that the provisions of ISO/IEC 90003:2014 *Software engineering – Guidelines for the application of ISO 9001:~~2008~~ 2015 to computer software* or an acceptable International, National or naval standard, are incorporated. The plans are to define responsibilities for the lifecycle activities, including verification, validation, module testing and integration with other components or systems.

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